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**SUMMARY OF A DESIGN COMPARISON OF CESIUM AND POTASSIUM  
AS WORKING FLUIDS IN INTEGRATED  
RANKINE CYCLE SPACE POWER PLANTS**

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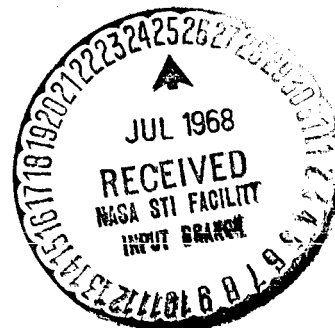
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MARCH 1968

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Oak Ridge, Tennessee  
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U.S. ATOMIC ENERGY COMMISSION

FOREWORD

This report summarizes an analytical comparison of cesium and potassium as working fluids for Rankine cycle space power plants. The work was conducted by the Oak Ridge National Laboratory for NASA under AEC Interagency Agreement 40-98-66, NASA Order W-12,353 under the technical management of A. P. Fraas of the Oak Ridge National Laboratory. Project management for NASA was performed by S. V. Manson of NASA Headquarters.

Acknowledgment

This report summarizes material presented in 12 reports covering the detailed facets of the problems investigated. The principal investigators were D. W. Burton, D. L. Clark, B. Cox, J. H. DeVan, B. H. Fitzgerald, A. G. Grindell, H. W. Hoffman, D. H. Jansen, R. L. Klueh, M. E. LaVerne, T. T. Robin, J. J. Tudor, L. V. Wilson, and H. C. Young of ORNL; D. W. McLaughlin and R. L. Thorkildsen of MTI; and W. D. Puchot, C. M. Glassmire, and T. C. Varljen of Westinghouse Astronuclear. In addition, much helpful advice and assistance were given by R. L. Cummings, A. J. Glassman, T. A. Moss, W. L. Stewart, and R. N. Weltmann of NASA Lewis Laboratory, J. J. Lynch of NASA Headquarters, and R. E. MacPherson of ORNL.

## CONTENTS

	<u>Page</u>
Abstract.....	1
Introduction.....	1
Summary.....	2
Delineation of Basic System.....	5
Flow Sheet.....	6
Full Power Design Conditions.....	8
Layout Studies of the Integrated Systems.....	10
Comparison of Cesium and Potassium Vapor Systems.....	13
Turbine-Generator.....	15
Thermodynamic and Aerodynamic Design.....	15
Turbine Bucket Erosion.....	19
Rotor Creep.....	24
Rotor and Bearing Dynamics.....	25
Thermal Stresses.....	26
Generator.....	28
Boiler.....	29
Condenser.....	29
Radiator.....	30
Pumps.....	30
System Control.....	32
Reactor Circuit Temperature and Flow Control.....	38
References.....	43

SUMMARY OF A DESIGN COMPARISON OF CESIUM AND POTASSIUM  
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A. P. Fraas

Abstract

The report summarizes a series of studies on the relative advantages and disadvantages of cesium and potassium as working fluids in Rankine cycle space power plants designed for an output of 300 Kw(e). The design studies, operating experience, and related information available on space power plants were reviewed to provide the basis for a consistent set of design precepts, and using these precepts designs were evolved for the turbine-generator, boiler, condenser, radiator, and pumps. These detailed studies have been covered in a series of twelve reports. This report summarizes the component designs from those detailed studies and presents the integrated systems that emerged from power plant layout studies. The latter included consideration of regenerative feed heating, the use of different types of pump, and system control problems.

A few significant differences between the two working fluids were found. Cesium gives a substantially smaller, lighter, simpler turbine, but requires a somewhat larger generator and boiler. The combined weight of these three components is about 285 lb lower in the cesium system than in the potassium system. There is little difference in the feed pump weights if a free turbine-driven feed pump is employed. If helical induction electromagnetic pumps are employed, the weight increment associated with the pump and related equipment is about 1657 lb greater for cesium than for potassium. For the case in which electromagnetic pumps are used the system weight for cesium is 7342 lb as compared to 5938 lb for potassium.

INTRODUCTION

A number of authorities have pointed out that the thermodynamic properties of cesium afford the turbine designer some degrees of freedom that are not available with potassium, and that these may make possible lighter, simpler, more reliable turbines. The problems involved have been examined by a number of organizations.<sup>1-3</sup> Some have concluded that there is little difference between the systems, whereas others have concluded that there would be a major advantage to the use of cesium. The Oak Ridge National Laboratory was asked by NASA to undertake a comparative study of the two systems with the objective of highlighting the principal differences that result from the use of one fluid or the other,

and the principal advantages and disadvantages of each from the standpoint of the design and development of the individual components and the complete integrated systems (AEC Interagency Agreement 40-98-66 NASA Order W-12,353).

The major problem areas involved in the design of the principal components such as the turbine-generator, boiler, condenser, radiator, pumps, etc., have been investigated, and a report covering each of these studies including recommended reference designs has been prepared.<sup>4-15</sup> This summary report is concerned with the integration of the various components to give a complete system that will satisfy as well as possible the many requirements that it should meet. As in the subsidiary studies on the individual components, a common set of design precepts has been employed to highlight as well as possible the relative advantages and disadvantages of the two working fluids.

#### SUMMARY

The size of the power plant as a whole, and the size and weight of most of the components in the system, are relatively independent of whether cesium or potassium is employed as the working fluid in the Rankine cycle. The overall thermodynamic cycle efficiency for operation between a given set of temperature limits is the same, and hence all of the components in the primary reactor circuit and in the radiator circuits are the same irrespective of the choice of working fluid. There are some differences in the boilers and condensers but these differences are not great. The only components in which there are large differences between the two working fluids are the turbine-generator unit and the electromagnetic feed pumps.

The much larger molecular weight of cesium as compared to potassium makes it possible to employ a greater temperature drop per stage with a lower turbine wheel tip speed while yet maintaining a high aerodynamic efficiency in the turbine. This makes it possible to reduce the number of turbine stages in the cesium turbine to roughly half the number required for a potassium vapor turbine, and this in turn makes it practicable to mount the turbine rotor directly on the end of the generator



rotor to give a compact turbine generator unit with only two bearings instead of the usual four-bearing configuration having a flexible coupling between the turbine and generator. For the reference designs of this study the resulting saving in turbine weight was estimated to be 560 lb. The large temperature drop per stage in the cesium turbine, however, does have the disadvantage that it gives a high axial temperature gradient in both the turbine rotor and the turbine stator, and this leads to major thermal stress problems. While it was not possible to obtain a definitive answer to these problems in the present study, preliminary analyses indicate that acceptable solutions probably can be found. Assuming that this can be done, a larger temperature drop per stage leads to lower wheel operating temperatures, and this, coupled with the lower stresses in the lower tip speed cesium turbine rotors, greatly eases the creep problem in the cesium rotors. In addition, analyses of the erosion problem indicate that the threshold for damage will exceed the design tip speed by a greater margin for cesium than for potassium.

It is difficult to assess the possible differences in reliability between the cesium and potassium turbines. The reliability advantages of the small two-bearing machine that would be possible with cesium must be weighed against the thermal stress and thermal distortion problems inherent in the use of a large temperature drop per stage.

The lower turbine speed desirable for use with cesium leads to a lower generator speed and a somewhat larger and heavier generator than would be the case for potassium. For the 300 kw electrical output unit of this study, the increase in generator weight appears to be of the order of 140 lb, which offsets about one-fourth of the 560 lb weight advantage of the small cesium turbine.

If a free turbine-driven feed pump is employed, the same characteristics of cesium that yield a smaller turbine for the turbine generator unit also lead to a smaller and lighter turbine driven centrifugal feed pump. The free turbine-driven feed pump weighs only 72 lb for potassium and 35 lb for cesium, a factor of two difference for the pump itself. However, the greater volume flow rate and density for the cesium lead to a greater vapor bleed requirement so that the overall weight penalty associated with the feed pump is 174 lb for cesium as compared to 135 lb for potassium. If an electromagnetic feed pump were employed, the much poorer electrical conductivity and higher weight flow rate associated with

cesium lead to an estimated pump weight of 1430 lb for cesium as compared to 372 lb for potassium. The difference in system specific weight after allowances for auxiliary equipment and electrical power requirements is even larger; the potassium pump is estimated to entail a total increment in system weight of 635 lb as opposed to 2292 lb for the cesium pump.

The helical induction electromagnetic pumps have no rotating parts so that the pump itself should be more reliable than a free turbine driven pump. However, electromagnetic pumps depend on a complex chain of switch gear, power supply, and cooling system equipment, so the overall system reliability may be lower than for the free turbine-driven feed pump.

The overall system weight not including the reactor and shield assembly, instrumentation and controls, or power conversion equipment (all of which should be the same irrespective of whether cesium or potassium is employed), was estimated to be 5185 lb for cesium or 5477 lb for potassium if a free turbine-driven feed pump is employed, and 7342 lb for cesium or 5938 lb for potassium if a helical induction electromagnetic feed pump is employed.

A survey of the operating experience with boiling cesium and potassium systems has shown that a total of 6000 hr of operating experience has been obtained with cesium systems as compared with about 136,000 hr of operating experience with potassium systems. All of the cesium systems that have been run were small, simple test loops whereas at least seven of the potassium systems were fairly large and complex, and included turbines, boilers, and condensers that were fairly good preliminary approximations to units that might be employed in a space power plant.

While there are gaps and some inconsistencies in the data, it appears that there is no significant difference between cesium and potassium from the standpoint of corrosion and mass transfer in refractory metal systems, and both fluids should be satisfactory.

A survey of the physical property, heat transfer, and fluid flow data on cesium and potassium indicates that, while more data would be desirable, sufficient data are available for both fluids to permit a meaningful comparison.

There appears to be no significant difference between the cesium and potassium system reference designs from the standpoint of system stability and control other than the lower melting point of cesium (83°F vs 145°F for potassium) which may affect some start-up problems.

#### DELINEATION OF BASIC SYSTEM

The original NASA work statement specified that the study should be directed toward a 300 Kw(e), 3-loop system with a lithium primary circuit, a Rankine cycle power conversion system using either potassium or cesium, and a set of parallel tertiary loops employed to transport the heat from the condenser to the radiator. The turbine inlet temperature was specified as 2150°F with 25°F of superheat, and the condenser temperature as 1330°F. The design life was specified as 40,000 hr. In subsequent discussions, ORNL was requested to perform additional calculations to show the effects of dropping the turbine inlet temperature to 2000°F, and of dropping the condenser temperature to 1200°F and 1040°F, respectively (see Table 1). Performance calculations were carried out for these additional operating conditions, but detailed design studies of components were made only for the single original set of design conditions.

Several refractory alloys were considered for use in the reference designs. Their relative merits were examined, particularly with respect to creep stress considerations, and it was decided that, for purposes of the subject study, D-43 (Nb-9 W-1 Zr)<sup>16</sup> would be employed for all elements of the structure that would require welding, and that TZM, a molybdenum alloy, would be used for the turbine rotor. While other alloys such as T-111, a tantalum alloy, might be used,<sup>17</sup> it was felt that the choice of other candidates would have little or no effect on the relative attractiveness of cesium and potassium. Similarly, it was felt that there would be important advantages to the use of stainless steel in the radiators,<sup>18</sup> and that its use there in the place of D-43 would have little effect on the relative merits of the two working fluids.

The work statement specified that ORNL should not do any design work on the reactor and shield assembly but should direct the study to the rest of the system.

Table 1. Thermodynamic Cycle Conditions

Fluid	Turbine Inlet			Condenser	
	Pressure (psia)	Temperature (°F)	Superheat (°F)	Pressure (psia)	Temperature (°F)
Reference Designs					
Cs	314.6	2150	25	23.6	1330
K	214.3	2150	25	10.4	1330
Parametric Studies					
Cs	314.6	2150	25	12.1	1200
	314.6	2150	25	4.4	1040
	220.4	2000	25	23.6	1330
	200.4	2000	25	12.1	1200
K	214.3	2150	25	4.8	1200
	214.3	2150	25	1.5	1040
	141.6	2000	25	10.4	1330
	141.6	2000	25	4.8	1200

### Flow Sheet

The principle features of the system studied are shown in the flow sheet of Fig. 1. This differs from the flow sheet used in the SNAP-50 work<sup>18</sup> in that it includes a regenerative feed heater and a free turbine-driven feed pump.<sup>19</sup> The regenerative feed heater has been included because detailed studies of the thermodynamics and aerodynamics of the turbine-generator indicate that the regenerative feed heater makes it possible to increase the overall cycle efficiency by 7.5% and reduce the radiator size by 8.8%.<sup>6</sup> Further, it provides a convenient way to remove moisture from the turbine and thus reduces the possibility of turbine bucket erosion and should increase the turbine efficiency by minimizing moisture churning losses. It also eases boiler design problems by reducing thermal stresses in the vicinity of the boiler inlet.<sup>20</sup> These problems are discussed in more detail in the companion report on turbine design.<sup>6</sup>

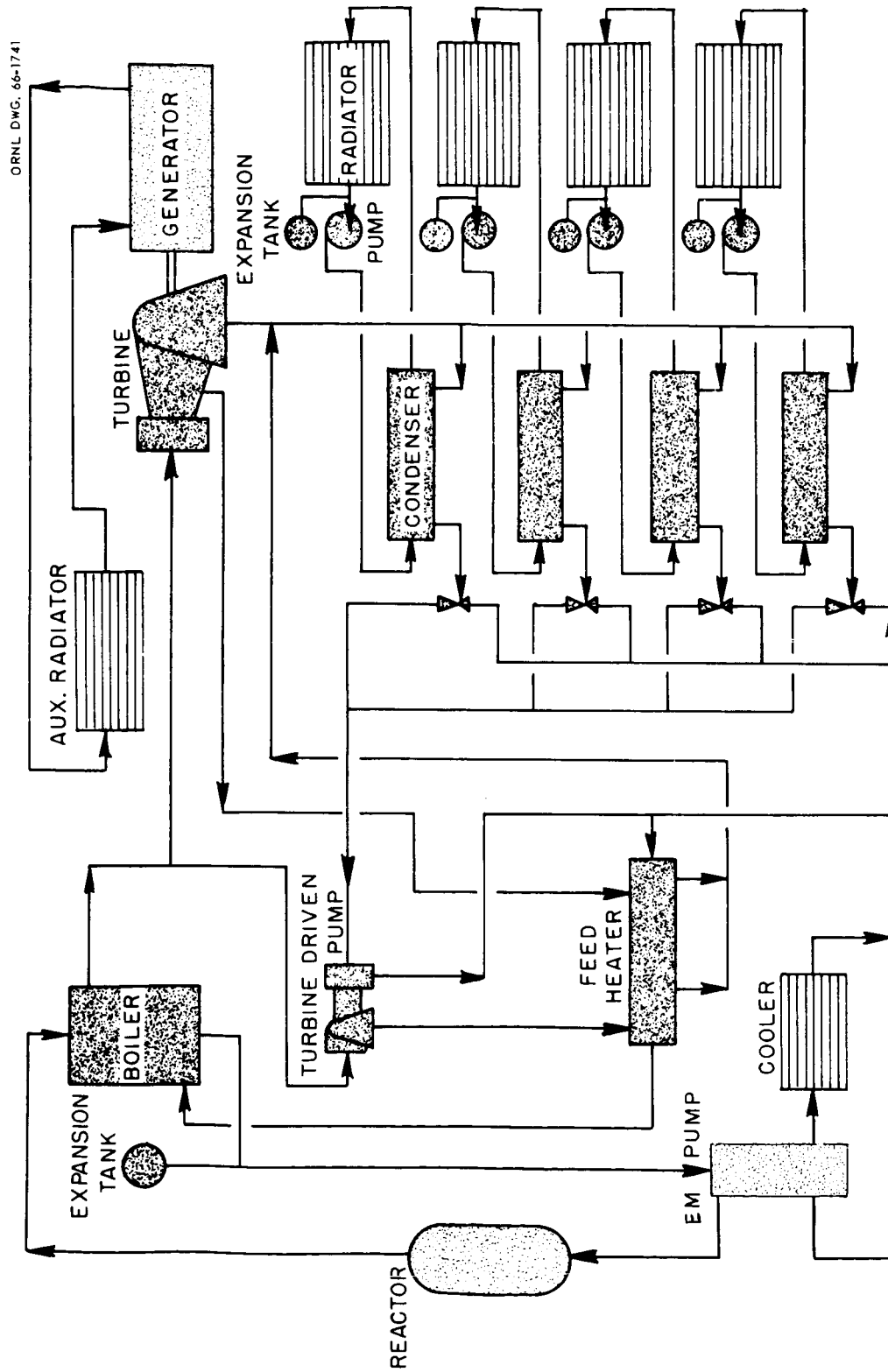


Fig. 1. Three-Loop System with a Liquid-Cooled Reactor Loop Heating the Boiler of a Rankine Cycle Loop Coupled to a Set of Parallel Indirect Radiator Loops.

A free turbine-driven feed pump rather than an electromagnetic pump has been employed in the layout studies for several reasons. The pump itself is much lighter and the loss associated with the power required to drive it is half as great as it would be for an electromagnetic pump. Most important of all, however, it appears to be more reliable than an electromagnetic pump because it does not depend on a chain of switch gear and auxiliary power supplies to keep it in operation. Further, tests of a 360 Kw(t) Rankine cycle system<sup>19</sup> have shown that a free turbine-driven feed pump can be employed with little or no control equipment, thus further increasing the reliability and reducing the weight of auxiliary equipment.

A fairly detailed set of design studies in the companion report on pumps indicate that helical induction electromagnetic pumps appear to be the best choice for the lithium and NaK circuits.<sup>10</sup> While canned rotor centrifugal pumps would probably be lighter, they introduce somewhat greater problems, particularly in the lithium circuit, and hence were not used in the reference design system of this study. Jet pumps were chosen as the best means for scavenging the condensers and providing cavitation suppression head for the feed pumps.<sup>10,19,21</sup>

#### Full Power Design Conditions

The temperature drops chosen for the lithium and NaK circuits have important effects. The choice in any given system depends on trade-offs between the pumping power, the size and weight of the equipment, the overall thermal efficiency, and such factors as thermal stresses in heat exchangers. Most previous studies have made use of a temperature drop in the radiator circuit of about 100°F. Time did not permit a detailed optimization study, hence the representative value of 100°F was employed for the reference design systems. Similar trade-off considerations apply to the lithium circuit. However, since details of the reactor design were specifically outside the scope of the ORNL study, and since without these it was out of the question to make optimization studies of the lithium circuit, again a round number of 100°F was employed as representative of the range of values chosen in other optimization studies for the lithium circuit temperature drop.

## LAYOUT STUDIES OF THE INTEGRATED SYSTEMS

The information on component sizes and weights developed in the detail design studies covered in companion reports is summarized in Table 2. Also included in Table 2 are data on the sizes of the expansion tanks and piping which were developed in the layout studies of this report.

In preparing several layouts to investigate various arrangements of the components, it was assumed that the power plant would be used in an unmanned vehicle or would be installed with a reactor-crew separation distance of 300 ft or more. This makes it possible to place the boiler outside the shield and not have the 13 Mev betas from the radioactive decay of activated lithium give too serious a Bremsstrahlung gamma source. While some might favor burying the boiler in the shield, this does not appear to be necessary. In any event, the reactor and shield design were outside the scope of the present study.

A major question in the design of an integrated reactor shield and space power plant system is presented by radiation emitted from a highly asymmetric reactor shield and scattered from the radiator. Unfortunately, the problem is extremely complex, and very little information is available to guide the designer. However, it is believed that the best information available is that being developed in a shielding study under way at ORNL at the time of writing. Preliminary results from this work indicate that a reasonably good compromise in the power plant and shield design can be effected by separating the reactor shield assembly from the base of the radiator by 3 ft to 7 ft as shown in Fig. 2, and this approach was adopted for the purposes of the subject study.

Of the several layouts prepared, that shown in Fig. 2 seemed best from the standpoints of structural strength and integrity under launch accelerations and vibration, materials compatibility, provisions for differential thermal expansion, accessibility for maintenance (particularly during development test work), and minimization of system liquid inventory. The latter is important not only from the standpoint of fluid inventory weight, but also as a means of reducing the fluid circuit transit times and control system lags as well as the hazard potential associated with combustion of the alkali metals in a ground test facility. The

Table 2. Summary of Size and Weight Data for Major Components<sup>6,8-10</sup>

Component	Cesium System			Potassium System		
	Diam (in.)	Length (in.)	Weight (lb)	Diam (in.)	Length (in.)	Weight (lb)
Turbine	16	7	120	20	25	680
Generator	23	20	690	21	19	550
Boiler	14.85	27	235	7.8	53	99
Condenser <sup>a</sup>	4.6	33	80	4.0	37	72
Radiator (Main)	120	192	1353	120	192	1353
Radiator (Auxiliary)	120	23	162	120	23	162
Regenerative feed heater	2.5	50	25	2.5	50	25
Pumps:						
Primary circuit	8	16	378	8	16	378
Feed pump - Free turbine <sup>a</sup>	10	6	135	12	7	174
Feed pump - EM - bare	18	40	1430	11	28	372
- with auxiliaries <sup>a</sup>			2292			635
Radiator circuits <sup>b</sup>	8	24	1320	8	24	1320
Expansion tanks:						
Primary circuit	6	15	15	6	15	15
Boiler circuit	4	15	10	4	10	10
Radiator circuit <sup>b</sup>	6	30	120	6	30	120
Piping:						
Primary	2	190	32	2	190	32
Vapor line to turbine	2	120	20	2	120	20
Vapor line to condenser	3.4	80	88	3.2	80	88
Boiler feed piping	1.5	180	15	1.1	180	15
Radiator circuits <sup>b</sup>	1.5	310	112	1.5	310	112
Liquid						
Reactor circuit <sup>c</sup>			30			30
Power conversion system			35			12
Radiator circuits			210			210
Total Weight (Free turbine feed pump)			5185			5477
Total Weight (Electromagnetic feed pump)			7342			5938

<sup>a</sup>Weight includes allowances for auxiliary equipment and power required to drive the feed pump.

<sup>b</sup>Dimensions given are for one component (four are required); weights are totals for all four components operating in parallel.

<sup>c</sup>Not including lithium in reactor and shield.



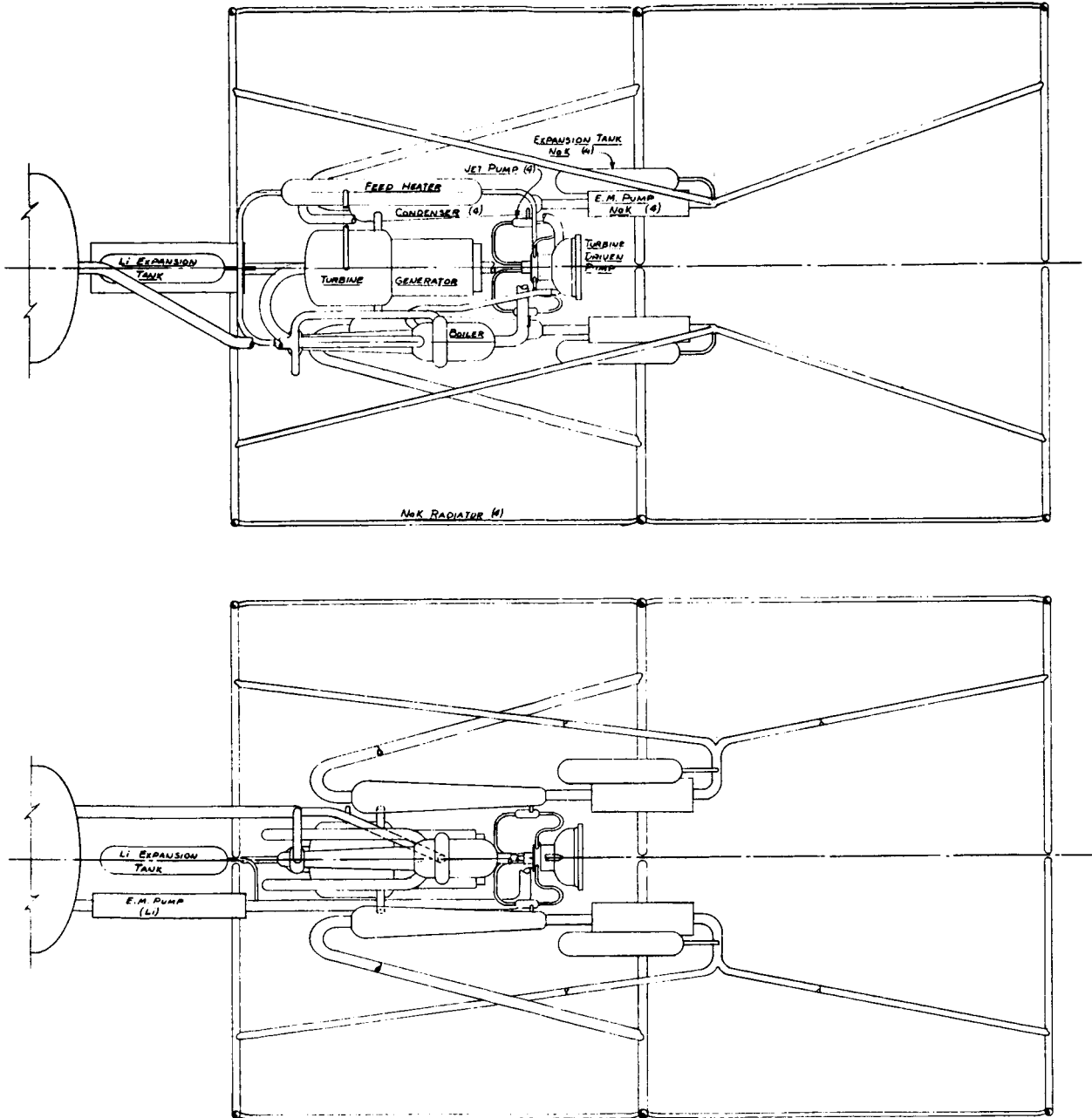


Fig. 2. Reference Design Power Plant Layout for the Potassium System.

latter is likely to be a major factor in determining the cost of such a facility.

The layout of Fig. 2 envisions division of the radiator into four quadrants that would be bolted together to provide a sturdy structure. Each quadrant would be served by a separate condenser, NaK circulating pump, and expansion tank. Each of the four condensers of these four systems would be scavenged by a jet pump that would return the feed to the main boiler feed pump.<sup>10,19</sup> In the event that any one of the four NaK circuits ceased to function, the condenser for that circuit would become ineffective, and vapor would tend to pass directly from the turbine outlet into the feed pump. However, the condenser scavenging jet pump would introduce such a severe restriction that the amount of vapor that could flow through the condenser into the feed line would be quite small. The precise effects on the rest of the system would depend on the amount of heat losses from various components and the elements of piping involved. It is believed that careful proportioning of these heat losses would make it possible to get satisfactory operation of the system without the use of a valve to isolate the condenser from the rest of the system. If this did not prove to be possible, it would be necessary to introduce a valve between the condenser outlet and the manifold joining the four condensers to the feed pump. If this were done, sufficient excess potassium would have to be carried in the expansion tank to permit loading up the inoperative condenser with liquid potassium. Provision for this has not been included in Table 2 or in the layout of Fig. 2. In principal, it would be possible to avoid the requirement for extra liquid inventory by providing a valve in the vapor line between the turbine and the condenser. In practice, the vapor line is so large that the weight of such a valve would be many times that of the extra liquid inventory, and it would be very hard - if not impossible - to assure that it would be truly leak tight.

#### COMPARISON OF CESIUM AND POTASSIUM VAPOR SYSTEMS

The effects of the choice of working fluid on the size and weight of the various components are summarized in Table 2 which was compiled from

the various detail design and layout studies.<sup>6-10</sup> Note that no weights are given for the reactor and shield assembly or for the instrumentation and control equipment and power conditioning equipment because these were outside the scope of the study, and, further, their size and weight would be independent of the choice of working fluid in the Rankine cycle.

Table 2 shows that the radiator size - and hence the power plant size - is essentially independent of the choice of working fluid, but the weights of some components differ substantially. The total weight of the components considered in the study is given at the bottom of Table 2 for two different conditions, that is, systems using a free turbine driven centrifugal feed pump and systems making use of helical induction electromagnetic feed pumps. In the former case the cesium system is 292 lb lighter than the potassium system whereas in the latter case it is 1124 lb heavier. As indicated in the report on pumps,<sup>10</sup> this stems from the relatively poor electrical conductivity, the higher volume flow rate, and the higher pump head associated with cesium as compared to potassium.

Except for the electromagnetic feed pump, the turbine-generator unit, and the boiler, other components have much the same weight for cesium as for potassium. The boiler is definitely lighter for potassium because the lower weight flow of the potassium appears to make possible designs that give higher power densities in the boiler than can be obtained with cesium for the same basic design boundary conditions.<sup>8</sup> On the other hand, the higher density of the cesium makes for a smaller turbine with fewer stages so that the reduction in turbine weight possible with cesium much more than offsets the increase in boiler weight.<sup>6</sup> However, the higher cesium molecular weight, and hence lower vapor sonic velocity, favor the use of a lower turbine rpm, and hence this in turn leads to a somewhat heavier generator.<sup>6</sup> The net effect of these various factors is to give a weight saving of about 260 lb for the cesium boiler-turbine-generator combination as compared to potassium. This difference represents about 5% of the total weight of plant components included in the study.

### Turbine-Generator

About half the total effort in the study was devoted to the turbine-generator unit, and many different aspects of the design were investigated in an effort to highlight the differences to be expected in well-proportioned units for the two working fluids.<sup>6</sup> In view of the fact that the size and weight of all the other components of the plant are directly related to the turbine efficiency, a high efficiency was considered a more important design objective than turbine size or weight. While reliability is an even more important consideration than efficiency, it is a far more difficult characteristic to evaluate analytically since it depends on subtle inter-relationships between such considerations as turbine bucket erosion, creep in the turbine rotor, thermal stresses in the rotor and casings, thermal distortion, and bearing lubrication. These problems were examined and an effort was made in each case to determine whether there would be any advantage to the use of cesium over potassium or vice versa.

### Thermodynamic and Aerodynamic Design

The first step in the turbine study was to carry out a parametric survey to determine the effects of the number of stages on the turbine rotor sizes and efficiencies using a common set of design precepts and allowances for aerodynamic, moisture churning, and seal leakage losses in the turbine. Figure 3 shows one of the most significant curves obtained from that study. Examination of the efficiency curves at the top of Fig. 3 indicates that a three-stage cesium turbine gives close to the maximum performance obtainable with cesium, and that a seven- or eight-stage turbine would give essentially the same performance with potassium. On the recommendation of Warner Stewart and A. J. Glassman of NASA-Lewis Lab, the three-stage cesium turbine and an eight-stage potassium turbine were chosen for primary reference design purposes. The layouts developed for these two cases are shown in Figs. 4 and 5. In addition, a two-stage cesium turbine and a five-stage potassium turbine were chosen as additional reference design cases to investigate the effects of balancing a small loss in efficiency against the simplifications in the turbine design

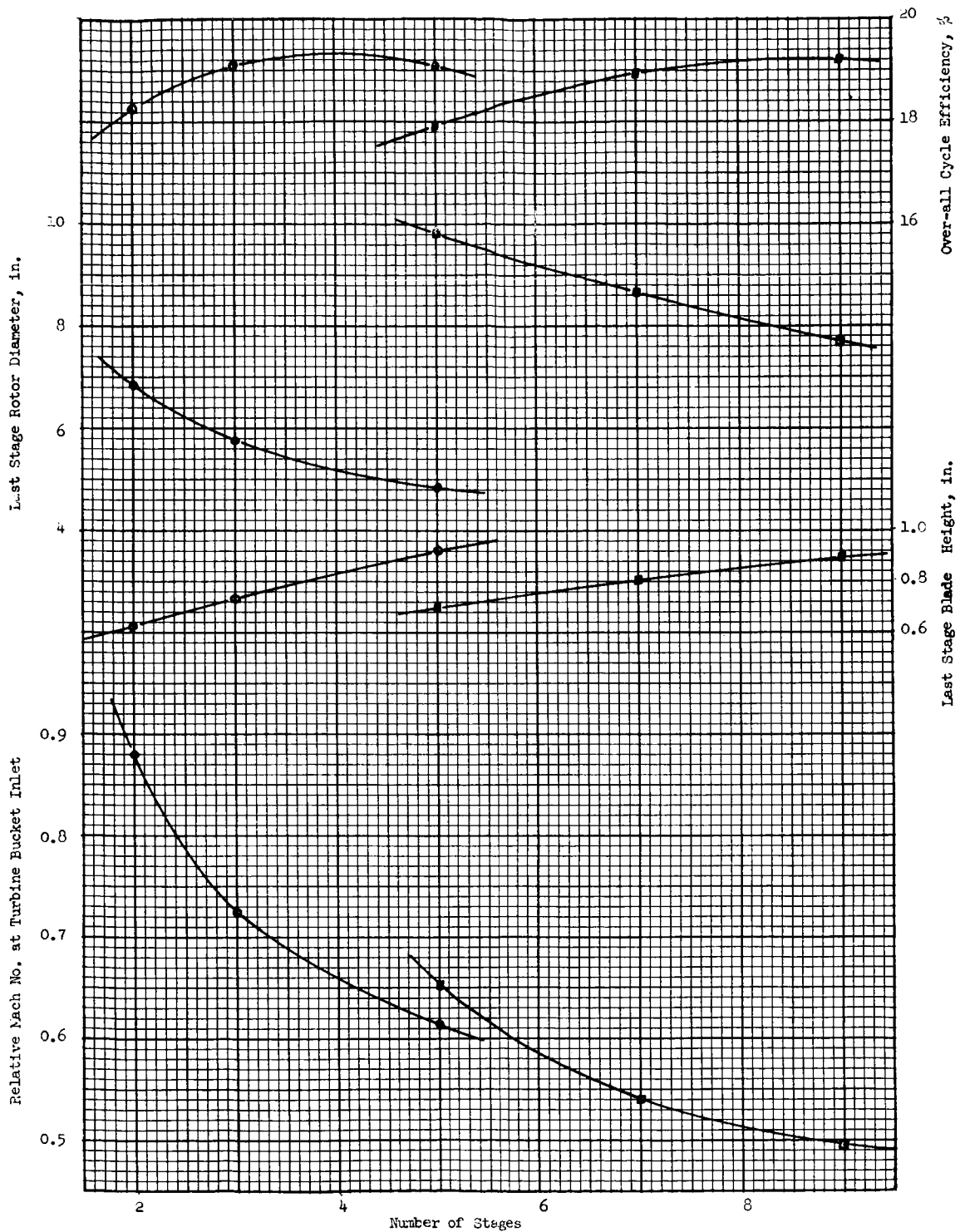


Fig. 3. Effects of Number of Stages on the Overall Thermal Efficiency, Size, and Inlet Mach Number of a Series of Cesium and Potassium Turbines Designed for 24,000 rpm with Allowances for Aerodynamic, Moisture, and Seal Leakage Losses with Regenerative Feed Heating.

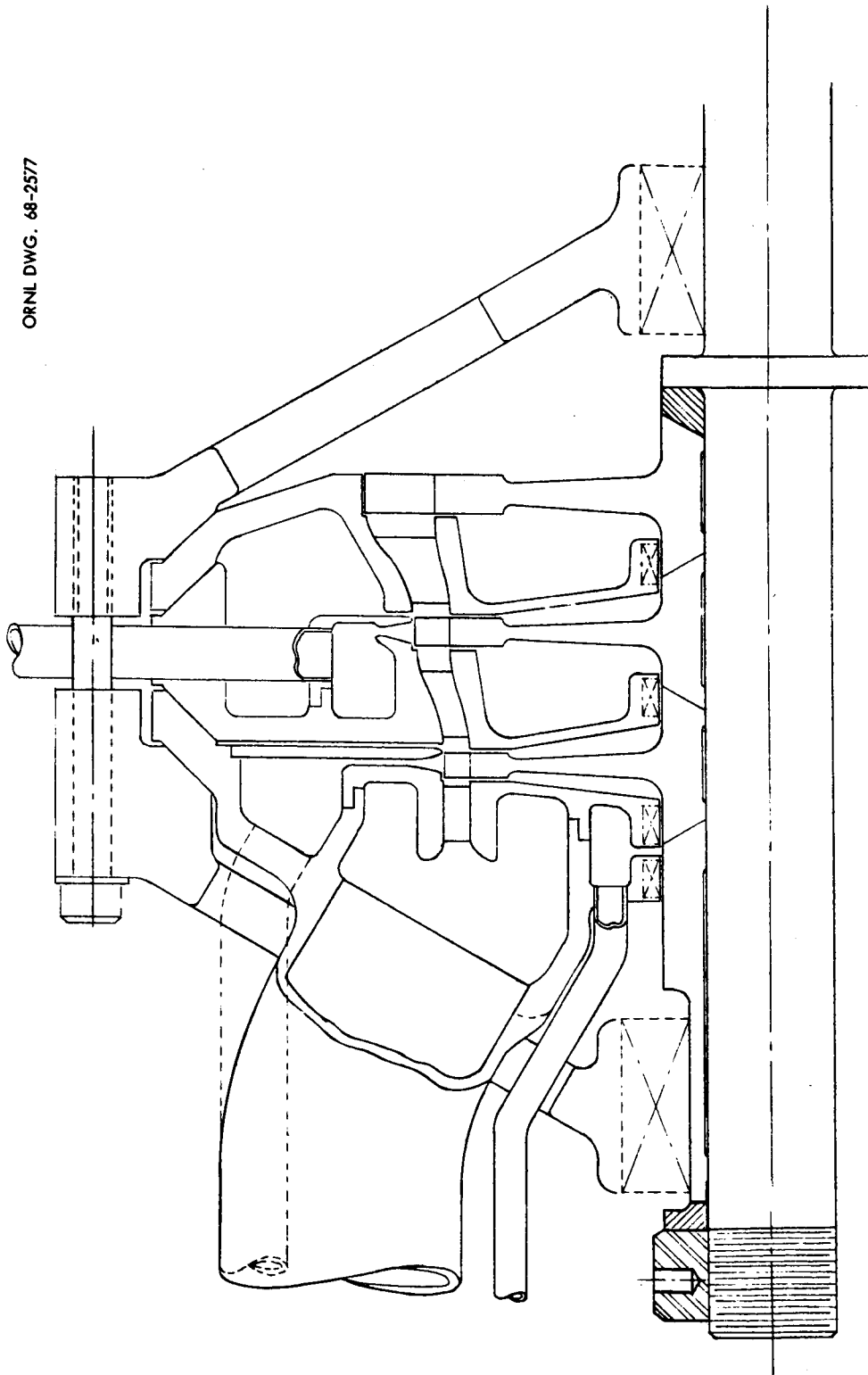


Fig. 4. Longitudinal Section through a Three-Stage Cesium Vapor Turbine Designed for a Four-Bearing Machine.

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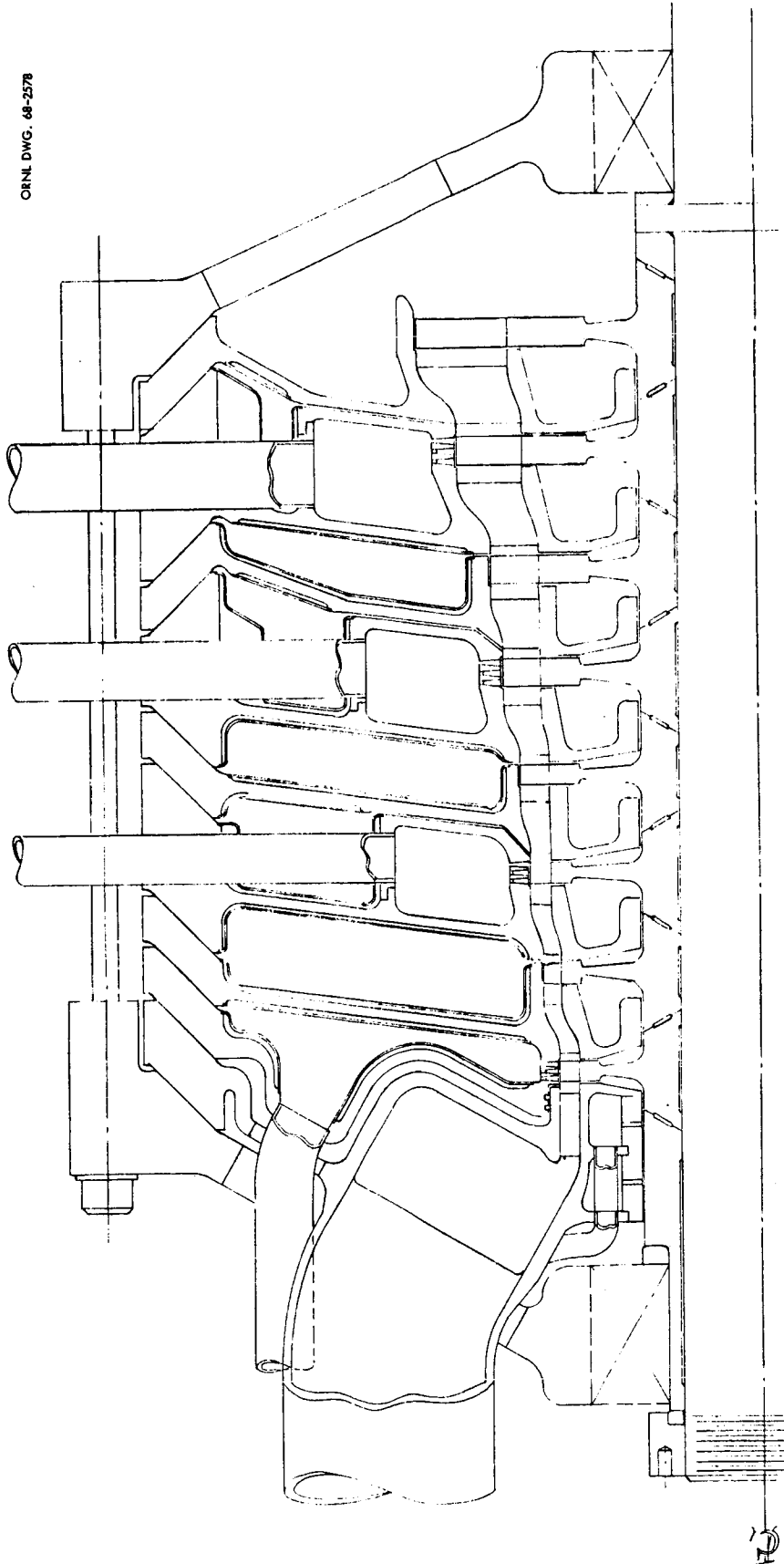


Fig. 5. Longitudinal Section through an Eight-Stage Potassium Turbine.

that the smaller number of stages would make possible, particularly in the potassium turbine. The latter yields a definite improvement in reliability by improving the rotor dynamics, easing bearing problems, etc. These problems were studied and the results are summarized below. The proportions of the reference design turbines are given in Table 3.

The effects of changing cycle design conditions on the overall cycle efficiency were also investigated and some of the results are summarized in Figs. 6, 7, and 8. These show the effects of varying the condenser temperature for two different turbine inlet temperatures with and without regenerative feed heating. From these curves it is easy to make rather good estimates of the effects on overall cycle efficiency of changes in the cycle design conditions.

The same techniques were employed for both potassium and cesium in carrying out the aerodynamic and thermodynamic analyses. While other analysts might have used somewhat higher or lower coefficients in estimating the various losses, the relative performance of cesium and potassium should be unaffected. The most important uncertainty in this work appears to the writer to be that associated with the thermodynamic calculations. There seems to be an anomaly in that the cesium and potassium idealized cycles with aerodynamic losses yield less than half a point difference in thermodynamic efficiency between the same temperature limits with no regenerative feed heating, but the difference increases approximately one point in favor of cesium with regenerative feed heating. These differences are increased by inclusion of moisture losses. They may stem from the thermodynamic data used.<sup>11,12</sup> In view of the small number of experimental physical property measurements on which the thermodynamic charts were based (for either cesium or potassium), the uncertainties in the absolute values, and the uncertainties in the empirical relations employed for interpolating and extrapolating the limited physical property data, it appears that this difference between cesium and potassium may be an artifact of the techniques used in constructing the thermodynamic diagrams rather than a real difference.

#### Turbine Bucket Erosion

A review of the turbine bucket erosion problem in wet vapor turbines disclosed that there is no widely accepted basis for estimating the



Table 3. Summary of Data for Reference Design Turbines

Case Number	1	2	3	4	5	6
Working fluid	Cs	Cs	Cs	K	K	K
Number of stages in turbine	3	2	3	5	8	3/2 <sup>a</sup>
Number of bearings	4	2	2	4	4	2
Average temperature of 1st stage rotor, °F	1740	1740	1880	1990	2050	1990
Rotor rpm	18,000	18,000	18,000	24,000	24,000	24,000
O.D. of 1st stage rotor, in.	5.923	7.215	5.923	6.297	5.076	6.297
Tip speed of 1st stage rotor, ft/sec	413	467	413	581	420	581
Tip speed of last stage rotor, ft/sec	597	578	597	943	838	943
O.D. of turbine casing, in.	16	16.7	16	20	20	20
Length of turbine casing, in.	9.6	6	7	15	25	16

<sup>a</sup>Three stages overhung at one end of generator and two at other.

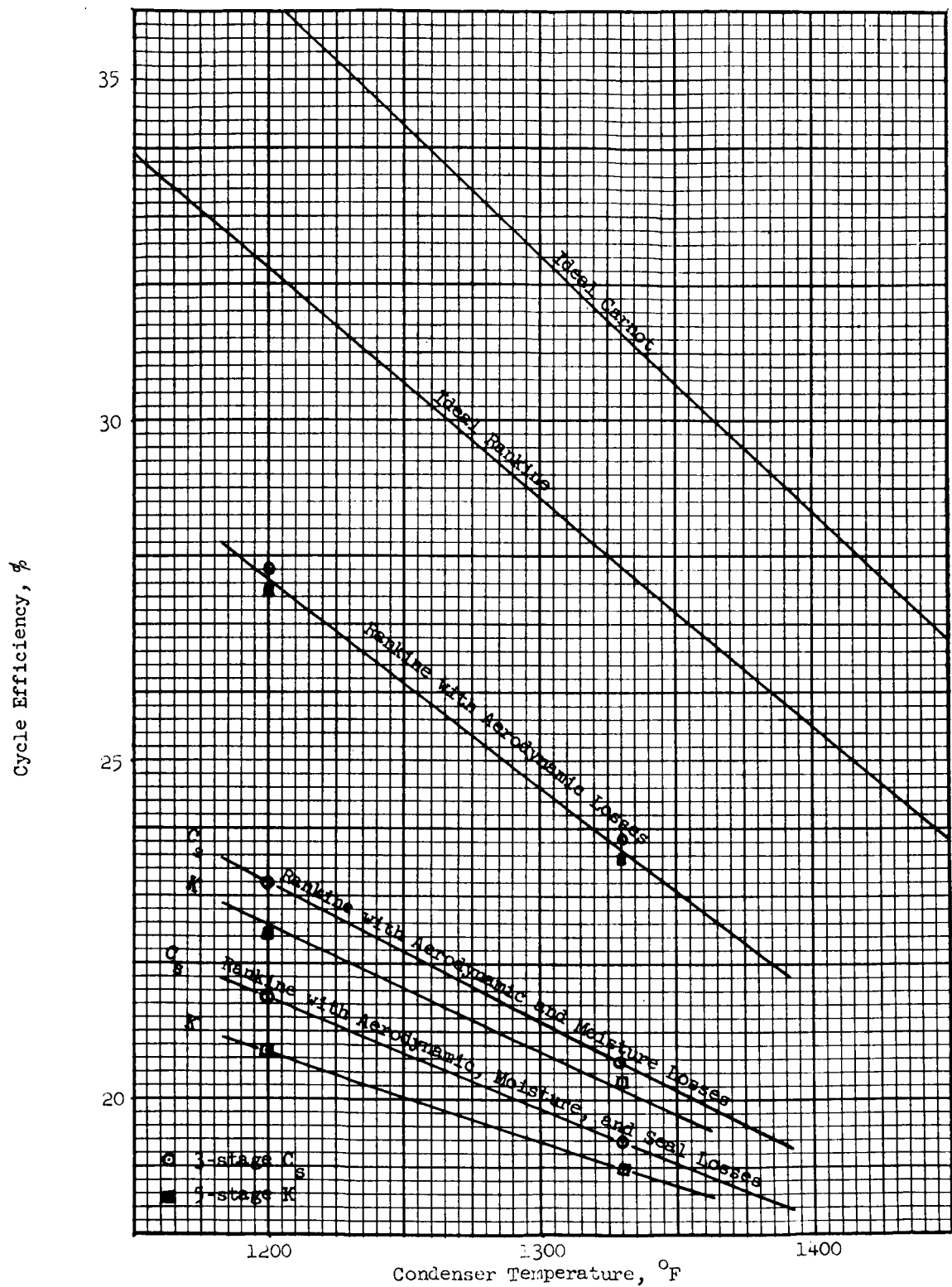


Fig. 6. Effects of Aerodynamic, Moisture, and Seal Losses on Cesium and Potassium Rankine Cycles with No Regenerative Feed Heating, a Turbine Inlet Temperature of 2150°F, and 25°F of Superheat.

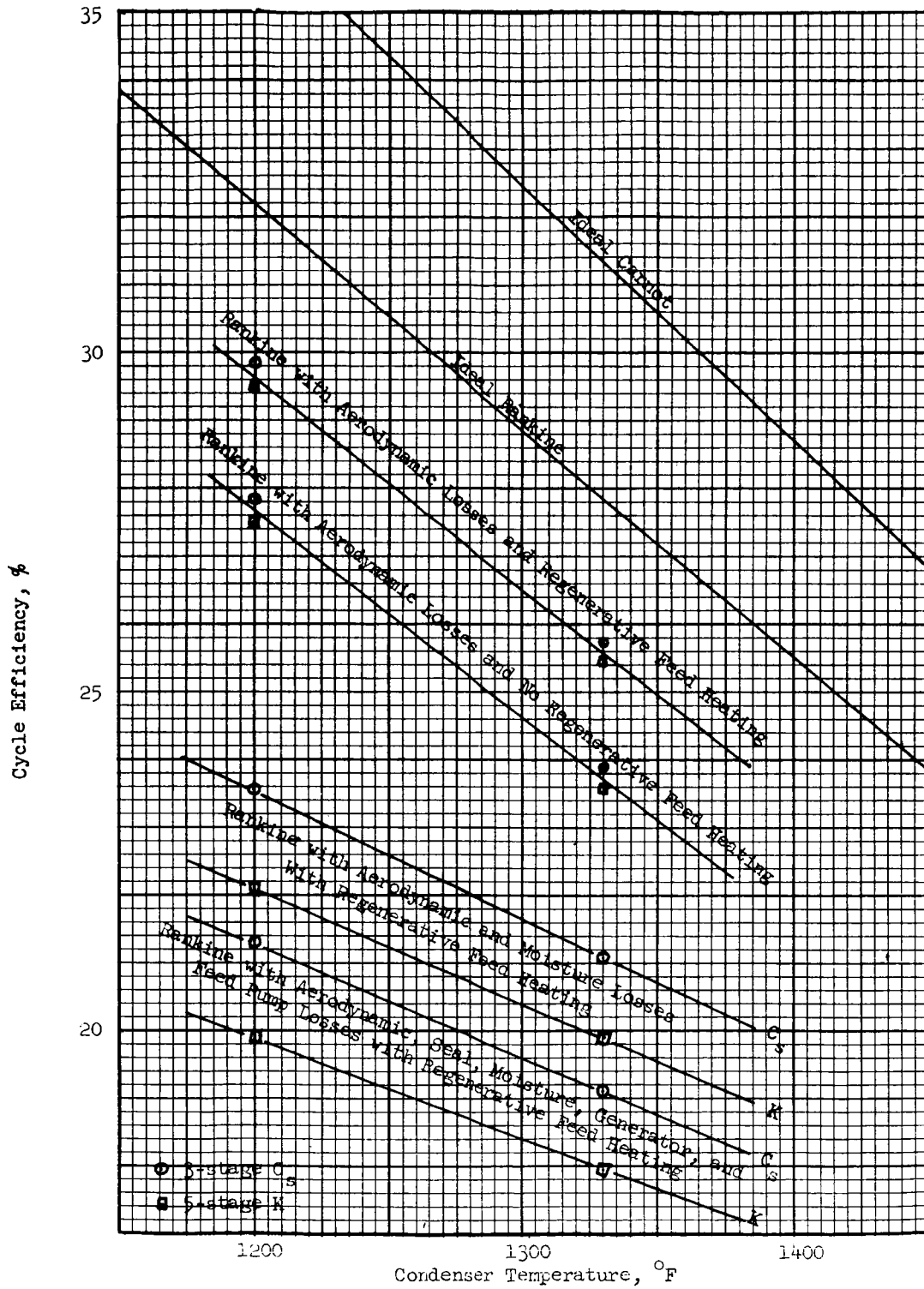


Fig. 7. Effects of Aerodynamic, Moisture, and Seal Losses on Cesium and Potassium Rankine Cycles with Regenerative Feed Heating, a Turbine Inlet Temperature of 2150°F, and 250°F of Superheat.

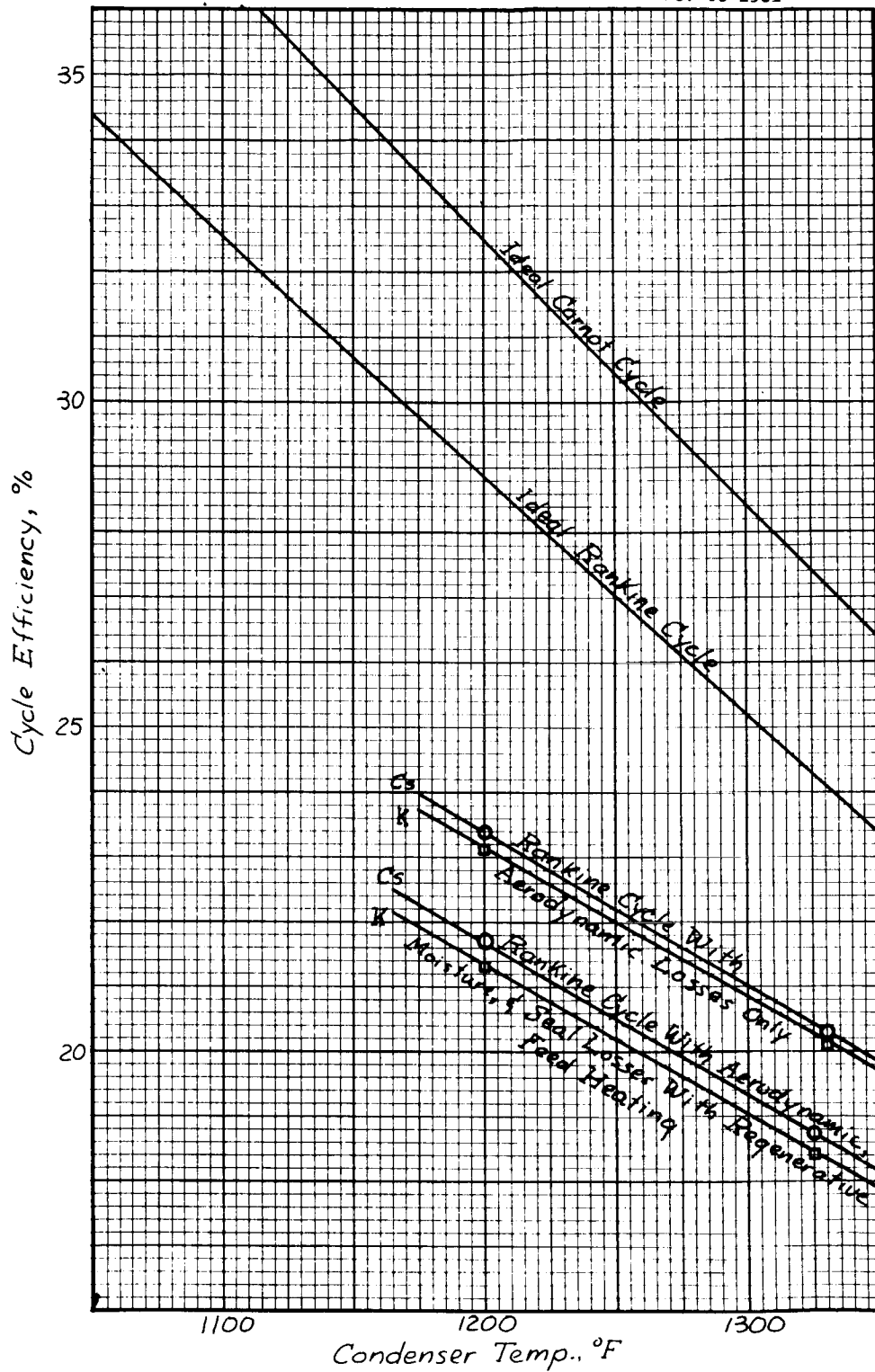


Fig. 8. Effects of Aerodynamic, Moisture, and Seal Losses of Cesium and Potassium Rankine Cycles with a Turbine Inlet Temperature of 2000°F as a Function of Condenser Temperature.

relative amounts of damage in the reference design turbines for the cesium and potassium vapor systems.<sup>7</sup> However, there are strong indications that turbine bucket erosion would be much less of a problem in cesium and potassium vapor turbines than in steam units and that cesium should present less of a problem than potassium.<sup>7</sup>

The best techniques for analyzing the erosion problem in wet vapor turbines appeared to be those developed at the Westinghouse Astronuclear Division under NASA contract, hence a subcontract was arranged with Westinghouse to estimate the rate of moisture formation and deposition, and, from this, the possibility of erosion in the four reference design turbines. The results of the study indicate that, if moisture removal between stages were employed, there should be no difficulty with turbine bucket erosion in either the cesium or potassium vapor turbines, and that the margin between design conditions and those that would induce erosion would be greater for cesium than for potassium for the reference designs studied.<sup>13</sup>

#### Rotor Creep

The efficiency of the first stages in the reference design turbines for both cesium and potassium is rather sensitive to the tip clearance because the blades are relatively short (see Figs. 4 and 5). Efficiency considerations make it necessary to limit the tip clearance to something of the order of 0.010 in. to 0.020 in., and hence it is important to design the rotor so that creep in the course of a 40,000 hr life will not use up more than a modest fraction of the initial tip clearance. An analysis of the problem was carried out at Mechanical Technology, Inc. under a subcontract. The results indicate that the reference design turbines for cesium are satisfactory from the standpoint of creep, but those for potassium are not.<sup>14</sup> If no changes were made in the turbine layouts, it would be necessary to reduce the turbine inlet temperature to the potassium turbines by about 150°F to keep the amount of creep within acceptable limits.<sup>14</sup> A more attractive course would be to modify the designs by reducing the diameter of the first stage rotor by about 22% to reduce the centrifugal stresses. This would cause a loss of about 3 points in the efficiency of the first stage, or less than half a point in the overall

efficiency of the turbine. This would reduce the overall cycle efficiency by about 0.2 point, which compares with the loss of about 1 point in the overall cycle efficiency that would be associated with dropping the turbine inlet temperature 100°F.

### Rotor and Bearing Dynamics

Two basic rotor and bearing configurations were considered for the reference design turbine-generator units of Table 3. In one, only two bearings were provided, one at either end of the generator rotor, and the turbine rotors were overhung from the end of the generator shaft. This is referred to here as a two-bearing machine. In the other configuration, the generator and turbine rotors would be individually straddle-mounted with one bearing at either end of the shaft of each unit with a flexible coupling between the two units to give a four-bearing machine. The two-bearing arrangement gives a smaller and lighter machine with no coupling problem but makes it necessary to integrate the turbine and generator very closely throughout the development program. The four-bearing machine gives greater flexibility in the development program and eases the heat loss and thermal stress problems associated with placing the hot turbine immediately adjacent to the relatively cold generator. Bearing alignment in the two-bearing machine would probably be easier to maintain than in the four-bearing machine, but it is difficult to attach quantitative values to any of the above considerations.

Increasing the number of stages in the turbine rotor, its rpm, and its diameter all increase the likelihood that difficulties may be encountered in the rotor and bearing dynamics. Of course it is desirable to keep the first critical speed above the operating range, but this is ordinarily not possible in the high speed units required for high specific output. In general, it has been found that satisfactory operating conditions can be obtained with the first two critical speeds below the operating range provided that the rotor layout and bearing design are such that damping in the bearings will limit the amplitude of the journal gyration in the bearings to substantially less than the bearing clearance so that the journal will not touch the bearing and no damage to the

bearing surfaces will occur in passing through the critical speeds in the course of startup or shutdown. The latter rotor and bearing dynamics problem is particularly complex if - as in this case - the bearings operate in the turbulent flow regime, and, to the writer's knowledge, only one computational technique has been developed to handle it for the conditions present in the reference designs. As a consequence, a sub-contract was arranged with the developers of this technique, Mechanical Technology, Inc., to carry out the analyses for the six reference design cases of Table 3. As can be seen in Table 4, the results of the study show that all six of the reference designs have their third critical speed above the design speed, and damping in the bearings appears to be adequate to limit to an acceptable value the amplitude of the journal gyrations in the bearings in passing through the first two critical speeds. The only marginal case appears to be the five-stage potassium turbine in which three stages would be overhung at one end of the generator shaft and two stages would be overhung at the other. Thus this latter configuration probably should be avoided.

It is also evident that it would be desirable to stiffen the rotor of the eight-stage potassium turbine by increasing the hub diameter somewhat to increase the third critical speed of that unit, but this does not appear to present any particular difficulties.

In appraising the results of Table 4 it appears that, from the standpoint of rotor and bearing dynamics, either a two- or a three-stage cesium turbine could be built with the rotor overhung from the end of the generator, thus eliminating the need for the coupling that would be required if a four-bearing machine were employed. This would also eliminate the possibility of troubles with bearing and rotor dynamics that might be induced by distortion and/or misalignment between the turbine and the generator in a four-bearing machine.

### Thermal Stresses

The temperature difference between the inlet and outlet of a turbine tends to induce thermal stresses in both the rotor and the stator. The magnitude of these stresses is directly proportional to the temperature gradient so that, as the number of stages in the turbine is reduced, the

Table 4. Summary of Data from Turbine Rotor and Bearing Dynamics and Creep Studies at MTI for ORNL<sup>14</sup>

Case Number	1	2	3	4	5	6
Working Fluid	Cs	Cs	Cs	K	K	K
Number of stages in turbine	3	2	3	5	8	3/2 <sup>a</sup>
Number of bearings	4	2	2	4	4	2
Average temperature of 1st stage rotor	1740	1740	1880	1990	2050	1990
Rotor rpm	18,000	18,000	18,000	24,000	24,000	24,000
O.D. of 1st stage rotor, in.	5.923	7.215	5.923	6.297	5.076	6.297
Tip speed of 1st stage rotor, ft/sec	413	467	413	581	420	581
Bearings:						
Journal diameter, in.	2	3	3	2	2	3
Journal length, in.	1.5	2.25	2.25	1.5	1.5	2.25
Journal temperature, °F	1330	830	830	1330	1330	830
Diametral clearance, in.	0.0030	0.0045	0.0045	0.0030	0.0030	0.0045
Approximate critical speeds:						
1st, rpm	7,000	5,500	5,500	6,500	6,500	5,800
2nd, rpm	14,000			15,000	15,000	8,300
3rd, rpm	>30,000	>30,000	>30,000	>40,000	>40,000	27,000
Maximum journal orbit amplitude: <sup>b</sup>						
1st critical speed, mils	0.01	0.005	0.006	0.01	0.01	0.01
2nd critical speed, mils	0.024			0.036	0.036	0.015
Design speed, mils	0.032	0.015	0.019	0.055	0.055	0.075
Friction horsepower, hp	4.0	8	8	4.4	4.4	12.6
Average tangential stress, psi	9,100	10,900	9,100	14,200	12,700	14,200
Rim growth (circumferential), in.	<<0.0027	<<0.003	<0.0027	0.0891	Fracture	0.0891

<sup>a</sup>Three stages overhung at one end of generator and two at other.

<sup>b</sup>Maximum amplitude occurred in generator rather than in turbine bearings in four-bearing machines.



thermal stresses tend to increase. There does not appear to be any appreciable difference between cesium and potassium turbines from the standpoint of thermal stress if the number of stages is the same, but it is clear that the thermal stress problems in a three-stage turbine are much more severe than those in an eight- or even a five-stage turbine. Work at KAPL<sup>22</sup> together with extensive experience at ORNL with components in liquid metal systems operating at temperatures above 1000°F indicates that it is desirable to avoid axial temperature gradients greater than roughly 100°F per diameter in cylindrical parts. As a consequence, in preparing the layouts for the reference design turbines, much thought was given to means for minimizing the thermal stresses, particularly in the turbines having two to five stages. The best approach appeared to be to segment the rotor and stator as indicated in Figs. 4 and 5 to provide heat dams between each pair of stages. Thus the rotor and stator were designed to consist of stacks of disks joined together by a large axial bolt through the center of the rotor and a series of small axial bolts around the perimeter of the stator.

In attempting to analyze the configurations of Figs. 4 and 5 it was found necessary to develop a new computer program to allow for the discontinuities represented by the parting surfaces between stages. Such a program was developed to obtain the temperature distribution, and a second program was adapted from an existing program to determine the thermal stresses in any given segment using the temperature distributions as input. The results of these analyses indicate that segmentation reduces the losses associated with axial heat flow and appears to keep the thermal stresses within acceptable limits in all of the reference design cases.<sup>15</sup>

A special set of thermal stress problems prevail in the turbine buckets. The static temperature of the high velocity vapor passing through any given stage determines the temperature of the bulk of the rotor in that stage. However, if the thermal conductivity of the rotor were low as compared to the local heat transfer coefficient, the temperature in the stagnation region along the leading edge of the turbine bucket might be expected to run close to the total temperature, the temperature of the vapor at the inlet to the stator of that stage. This would induce local

thermal stresses that would be directly proportional to the temperature drop across the stage.

Tests with thermocouples in high velocity gas streams normally yield indicated temperatures roughly midway between the static temperature and the total temperature of the gas stream because heat is conducted through a short path from the high temperature portion of the thermocouple perimeter to the low temperature portion. The problem is complicated in a wet vapor by the fact that droplets will tend to impinge on the leading edge and re-evaporate so that evaporative cooling may occur just downstream of the stagnation region. Attempts were made to analyze the problem, but there are so many uncertainties that any such analysis is open to serious question. However, it does appear that the local temperatures along the leading edge of the turbine buckets might run roughly midway between the total temperature and the static temperature, and that this would be likely to induce serious local thermal stresses in turbines in which a high temperature drop per stage is employed. If a two- or three-stage cesium turbine development program is contemplated, it appears that one of the first steps should be an experimental investigation of the local thermal stresses stemming from the high temperatures in the stagnation region along the leading edges of the turbine buckets.

#### Generator

The principal difference between the cesium and potassium turbines so far as generators are concerned is the desirability of using 24,000 rpm for the potassium turbine and 18,000 rpm for the cesium turbine. The lower speed eases the rotor and bearing dynamics problems but leads to a net increase in the size and weight of the generator rotor and stator. The problem was discussed with Westinghouse engineers at Lima, Ohio who were in the course of a study of generators for space power plants for the AEC. They kindly added two cases to the series used in their study and supplied the data used in Table 2. Note in Table 2 that the generator for the cesium system is 140 lb heavier than that for the potassium system.

### Boiler

Boiler designs were developed for both potassium and cesium using two different design approaches.<sup>8</sup> The first was based on the use of vortex generator inserts to centrifuge droplets to the tube walls and thus improve heat transfer in the mist flow region between the annular film flow portion of the boiler and the superheater. The second design approach pursued was directed toward designing for low liquid entrainment in the boiling region so that annular liquid film flow could be maintained up to qualities of 90% to 97%. Both straight and tapered tubes were considered in the latter approach together with the use of a combination parallel flow-counter flow configuration designed to reduce the heat flux in the transition region and thus help to defer the transition from annular flow to a dry wall-mist flow condition to a higher vapor quality.<sup>8</sup>

The results of the design studies indicate that the low liquid entrainment approach makes it possible to reduce the size and weight of the boiler appreciably over the vortex generator approach. The reduction in weight was estimated to be a factor of approximately two for potassium and about 20% for cesium. As a consequence the boilers designed by the low entrainment approach were chosen for the overall system reference design.

The higher weight flows associated with the use of cesium led to somewhat larger and heavier boilers for cesium than for potassium irrespective of whether the vortex generator or the low liquid entrainment approach was employed, but the differences in favor of the low liquid entrainment approach were much greater for potassium.<sup>8</sup>

### Condenser

Condenser designs were developed for both cesium and potassium using tapered tubes to maintain a fairly uniformly high vapor velocity down the tubes to sweep the liquid film toward the tube outlet under zero-g conditions.<sup>9</sup> As was found to be in the case in the boilers, a somewhat larger number of shorter tubes was required for the cesium condensers, but the overall difference in the weight of the components was almost trivial, the set of four cesium condensers weighing 188 lb per power plant as compared to 78 lb for potassium.<sup>9</sup>

### Radiator

A number of different types of radiator heat transfer surface were considered including the three finned tube configurations of Fig. 9 and several configurations employing heat pipes as fins.<sup>9</sup> The heat pipe configurations were dropped because several thermal stress problems appear to be inherent in any of the configurations that have been proposed in the literature. The finned configurations of Fig. 9 were compared and it was found that the use of a reflector as indicated in Fig. 9c yielded a weight about 40% lower than the more conventional configuration of Fig. 9b when the proportions of both were optimized for minimum specific weight, hence the thin tube and reflector configuration of Fig. 9b was chosen for reference design purposes.<sup>9</sup>

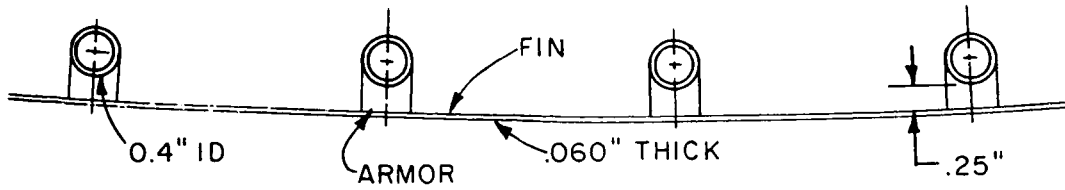
There should be no difference in the weight of the radiators or radiator systems irrespective of whether cesium or potassium is used because the overall thermal efficiency of the two thermodynamic cycles would be the same.

### Pumps

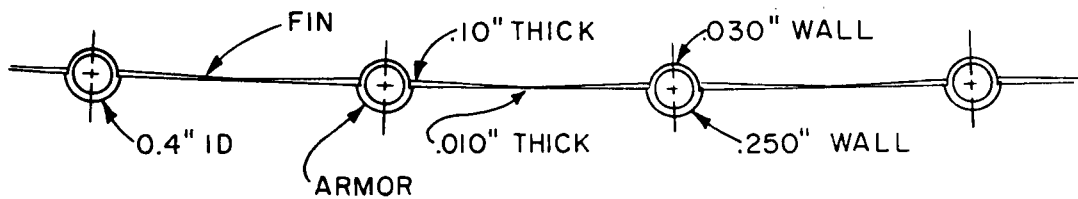
Various types of pump were considered for use in the three liquid systems of the power plant.<sup>10</sup> These included both ac and dc electromagnetic pumps, electric motor driven centrifugal pumps, and, for the boiler feed pump service, free turbine driven centrifugal pumps. Preliminary designs and weight estimates were prepared for each of the principal types of pump for each of the different types of service for which it was applicable using a common set of design precepts.<sup>10</sup>

The boiler feed pump presented a very complex set of problems. Studies of cavitation suppression head requirements coupled with system heat loss and radiator heat rejection considerations indicated that it would be best to employ jet pumps to scavenge the condensers and provide some cavitation suppression head at the inlet to the feed pumps rather than depend solely on subcooling. Further, the studies indicated that, to accomplish this, the flow through centrifugal pumps should be increased by 25% over the boiler feed requirements to provide the extra flow needed

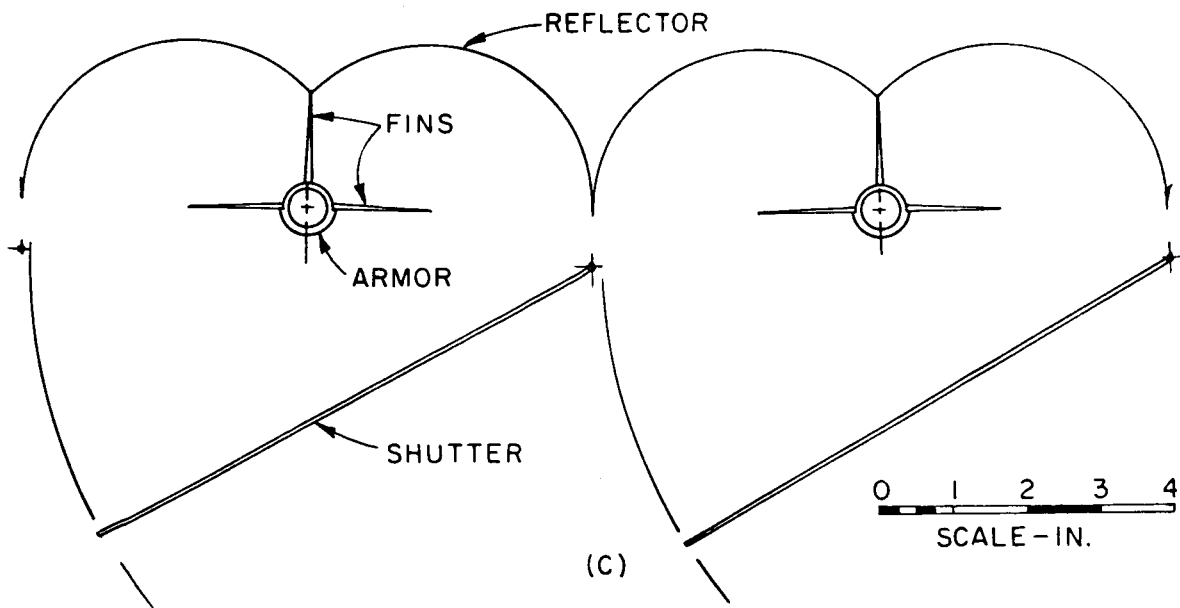
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(A)



(B)



(C)

Fig. 9. Three Typical Tube-Fin-Armor Configurations for Radiators for Space Vehicles.

to drive the jet pumps. Similarly, estimates indicated that the lower cavitation suppression head requirements of electromagnetic pumps gave a well-proportioned installation with the flow through the boiler feed pump increased by 12% to provide the fluid to drive the jet pumps.<sup>10</sup>

In comparing the various types of pump for boiler feed service it was found necessary to estimate not only the weight of the pump unit itself but also appropriate weight penalties to allow for the electric power consumed, power conditioning equipment, extra cooling system and radiator requirements, etc. In estimating these weight penalties a relatively small allowance for the power plant weight increment per unit of extra power plant capacity was chosen, that is, 10 lb/kw of electric power required for the pumps. The resulting values for the total incremental weight chargeable to the feed pumps are summarized in Table 2. These extra weight penalties were not included for the NaK and lithium pumps of Table 2 because they would be the same for cesium as for potassium. They are given, however, in the report covering the pump design study.<sup>10</sup>

#### SYSTEM CONTROL

An important consideration in the design of any power plant is its behavior under idle, startup, part-load, and transient conditions. The characteristics of the various components in the system are sufficiently complex that there is a wide-spread tendency to design the individual components on the basis of their full power design performance with no regard to part-load operation, couple them together with pipes, introduce a number of valves, and then call in instrumentation and control experts and ask them to provide a number of black boxes that will keep operating conditions within prescribed limits. This may lead to the conclusion that the only thing to do is to get the power plant from zero power to full power as quickly as possible, and then keep it at full power irrespective of the electrical load and simply dissipate unneeded electric power with an electrical resistance grid. This crude approach entails running the power plant at full power and hence at peak design temperatures throughout its life. This approach will cut the life of the power plant at least in half because, in a power plant designed to operate at as high a

temperature as possible, the creep rate in pressure vessels, turbine wheels, piping, fuel element capsules, etc., increases very rapidly with temperature. Perhaps even more important, the diffusion rates of fission products through the crystal lattice of any ceramic fuel increase rapidly with temperature, and this increases the rate at which both swelling of the ceramic fuel matrix and a buildup of pressure within the fuel element capsule occur. Further, operation at full power when only half power is required will double the rate of fuel burnup. This will inevitably shorten the life of the reactor. Thus there are important incentives to reduce the power and temperature for part-load operation. This, in turn, means that the individual components should be designed to facilitate control under part-load conditions.

It is not easy to evolve a set of design criteria for good system stability and control characteristics at part load. The only way to reduce the power output of a turbine running at constant speed is to reduce the density of the fluid flowing through the turbine. An obvious step is to reduce the turbine inlet pressure and temperature. Figures 10 and 11 show the turbine inlet temperature as a function of load for the cesium and potassium reference design turbines of this study. The curves are essentially similar and show the natural part-load requirements of a typical turbine if it is supplied with a constant amount of superheat. As indicated in the companion report on boilers, the characteristics of the boiler-superheater unit would cause the amount of superheat to increase somewhat as the load is reduced, but the curves shown in Fig. 10 or 11 represent a good approximation to the conditions that would prevail. In short, the vapor pressure and density in the turbine should be approximately directly proportional to the power so that the temperature will drop with reductions in power from the full design power condition.

Inasmuch as the turbine acts as if it were a series of critical pressure drop orifices, the pressure ratio across the turbine is naturally inclined to remain constant. The turbine outlet temperature, assuming this condition, is also plotted in Figs. 10 and 11 to indicate the ideal condenser temperature associated with any given boiler outlet temperature (i.e., any given turbine inlet temperature). It happens that the mean radiator temperature will drop more rapidly than the turbine outlet

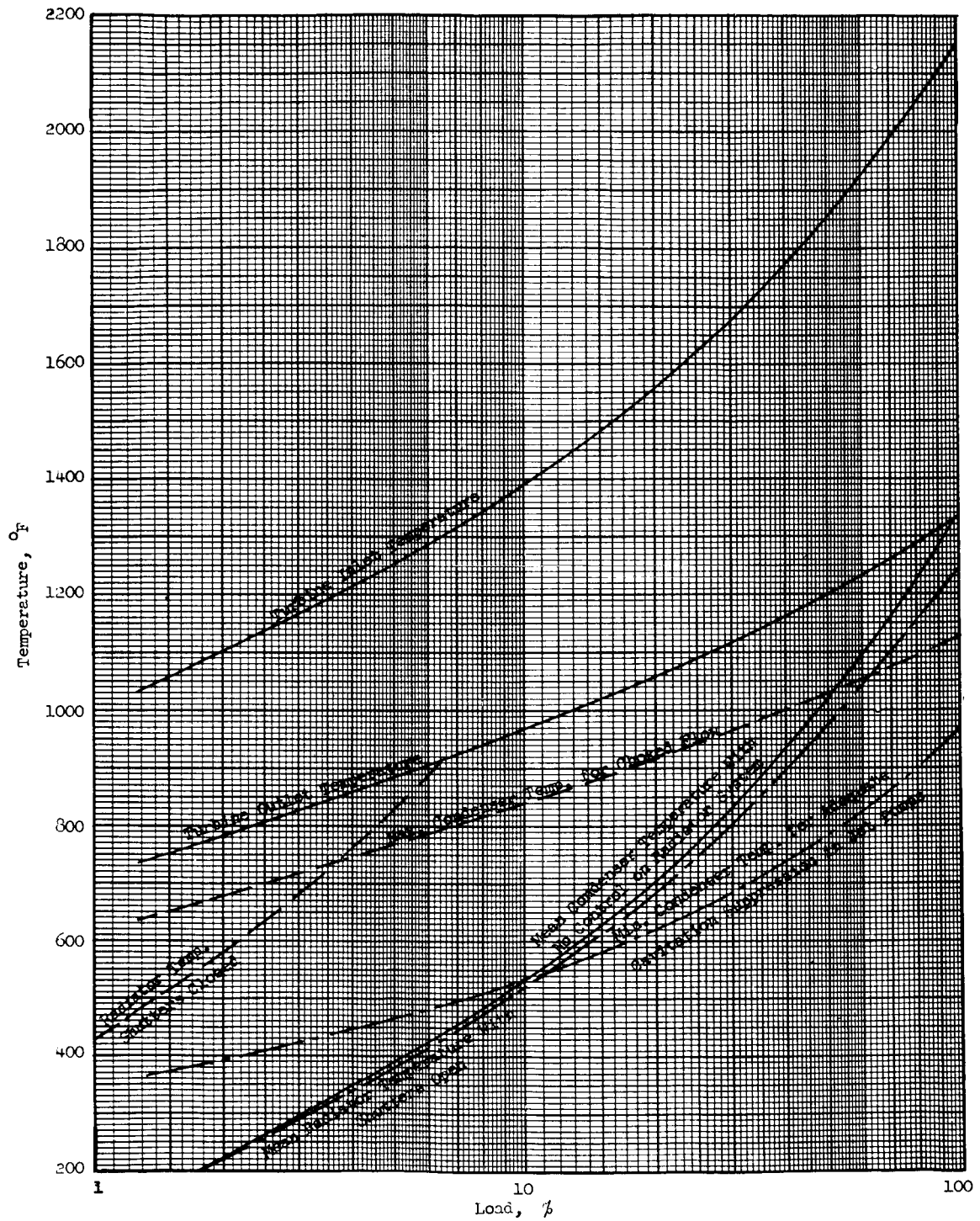


Fig. 10. Typical Temperatures at Representative Points in the Cesium System as Functions of Load.



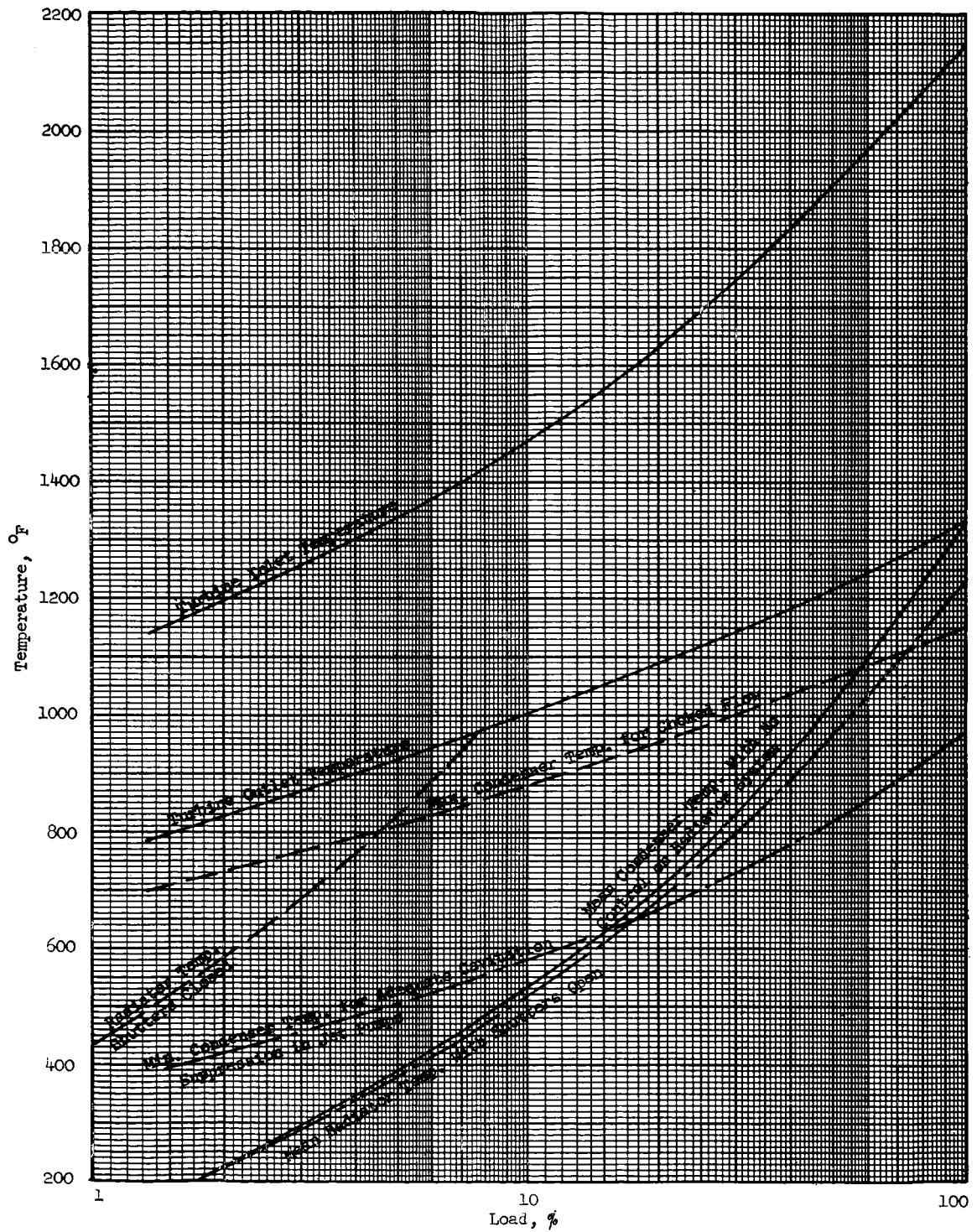


Fig. 11. Typical Temperatures at Representative Points in the Potassium System as Functions of Load.

temperature because the rate at which heat will be emitted from a space radiator varies as the fourth power of the absolute temperature of the radiator. After allowing for the temperature difference between the working fluid and the NaK in the radiator circuit, it is possible to work back from the average radiator temperature as defined for full power design conditions and delineate a condenser temperature as a function of load for operation with no control on the radiator system. Unfortunately, the resulting values are for a mean radiator temperature, and do not give a complete picture of the temperature structure within the condenser. The problem is complicated by the fact that the vapor density for both cesium and potassium falls off very rapidly with a reduction in temperature (see Fig. 12) so that choking will occur at the inlet to the condenser tubes if the system is designed properly. Space does not permit a more detailed discussion but it can be said that this is the best place for choking to occur as it will make it possible to maintain a uniform fluid flow distribution through the condenser.<sup>21</sup> If choking were to occur at some other point, a poor flow distribution in the condenser would be likely to result. For the condensers of the reference design power plants, the maximum condenser temperature for choking of the flow at the inlet to the condenser tubes is shown as a dashed line not far below the turbine outlet temperature.

The condenser temperature can be allowed to drop somewhat below that for choking of the vapor flow at the inlet to the condenser tubes. The system will still operate satisfactorily provided that the liquid can be scavenged from the condenser. This in turn means that there must be sufficient cavitation suppression head available from subcooling of the liquid leaving the condenser so that the pump scavenging the radiator will function satisfactorily. Previous ORNL experience indicates that the most effective way to scavenge liquid from a condenser at low condenser pressures is to employ a jet pump, and that such a pump can be designed to operate with a cavitation suppression head of around 2 ft at full flow, and as little as 0.01 ft at very low loads. To show the effect of this boundary condition an additional curve has been added in Figs. 10 and 11 to show the minimum condenser outlet temperature for which subcooling the

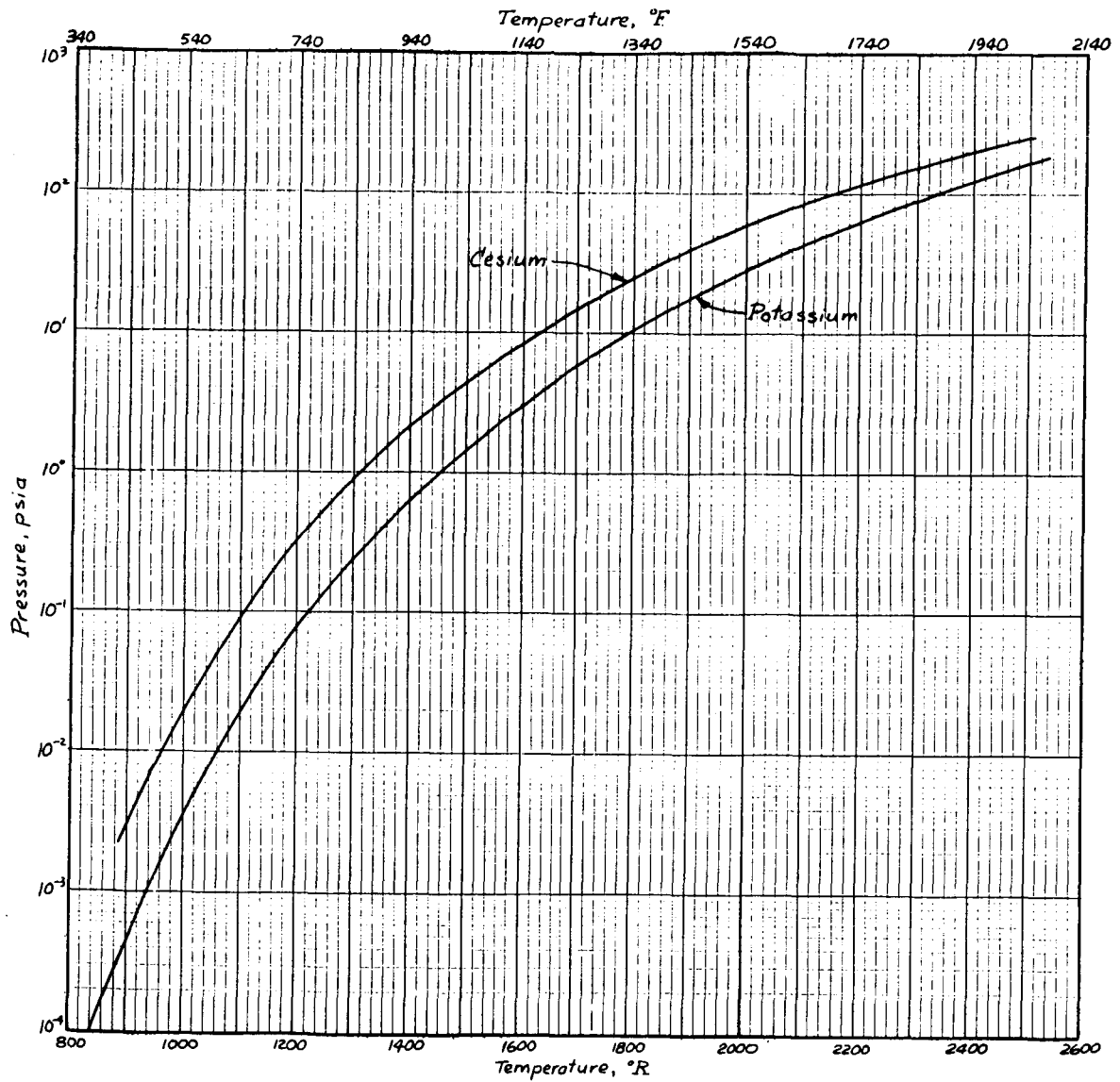


Fig. 12. Saturated Vapor Pressure as a Function of Temperature for Cesium and Potassium.

the condensate 100°F will suppress cavitation in a jet pump scavenging the condenser.

Inasmuch as it is desirable but not essential to maintain the condenser temperature somewhere between the turbine outlet temperature and the maximum condenser temperature for choked flow and it is essential to maintain the condenser temperature above the minimum for adequate cavitation suppression in the scavenging pumps, it is evident that some means should be provided to reduce the amount of heat lost from the radiator system at low loads. One approach to this is to make use of a by-pass valve in the NaK circuit so that a portion of the NaK would be recirculated through the radiator without passing through the condenser, and hence the mean NaK temperature in the radiator would be much below the mean NaK temperature in the condenser. This approach has the disadvantage of introducing a large temperature difference in the flow immediately downstream of the mixing valve. ORNL experience has shown that this in turn leads to severe thermal stresses and cracking in the walls of the passage.<sup>20</sup> Another approach is to employ shutters on the outside of the radiator as indicated in Fig. 9c. These can be opened and closed automatically by using an individual thermostat on each shutter.<sup>21</sup>

#### Reactor Circuit Temperature and Flow Control

The four principal manners in which the reactor circuit temperature and flow might be controlled are illustrated in Figs. 13 and 14. The first of these is shown at the left in Fig. 13. The reactor outlet temperature and flow rate would be held constant and the other temperatures of the system would vary as indicated. The line shown for the boiler temperature was prepared assuming a once-through boiler with no throttle valve between the boiler outlet and the turbine but with a valve or other device designed to control the rate at which liquid would be fed into the boiler. Note that this arrangement leads to an increase in the reactor inlet temperature as the load is reduced as well as an increase in the superheater outlet temperature. This arrangement would lead to very large temperature differences at the inlet to the boiler, which would mean both severe thermal stresses and unstable boiling conditions.<sup>8</sup>

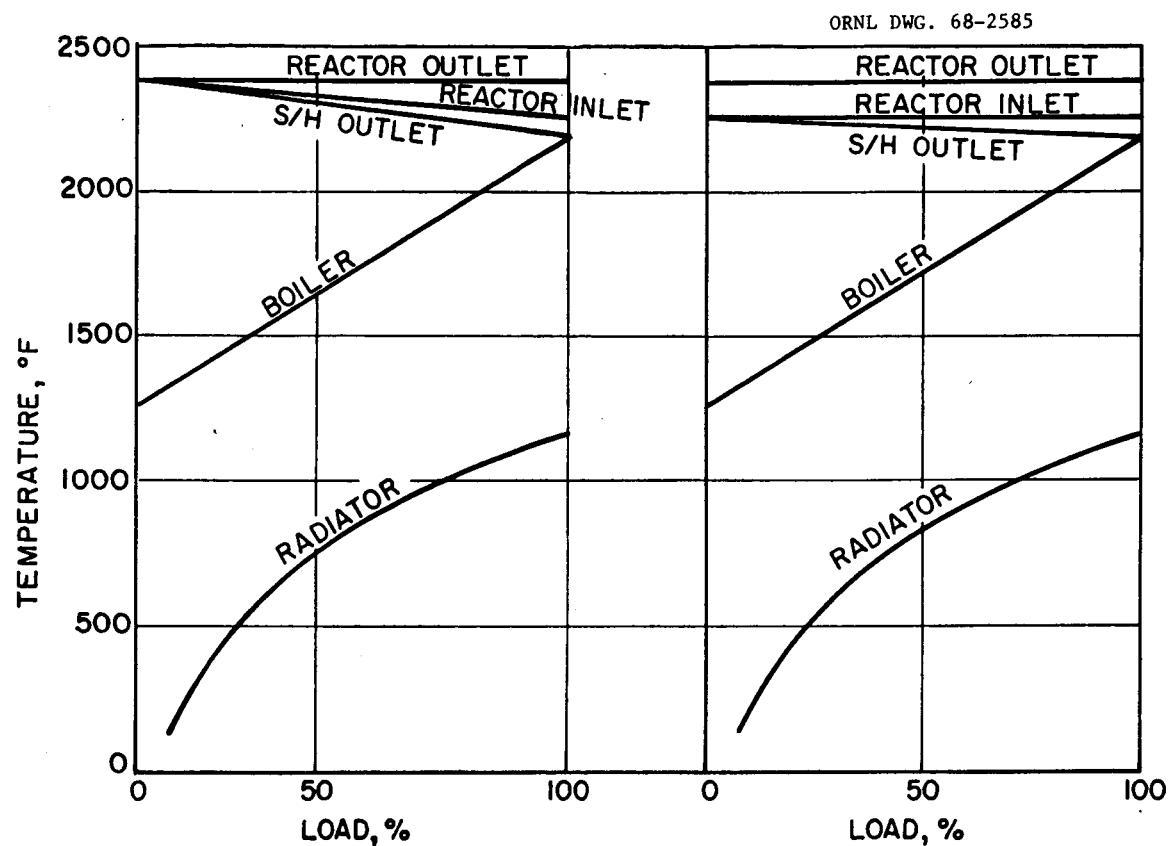


Fig. 13. Effects of Load on System Temperatures in which the Reactor Circuit Temperature and Flow May Be Controlled.

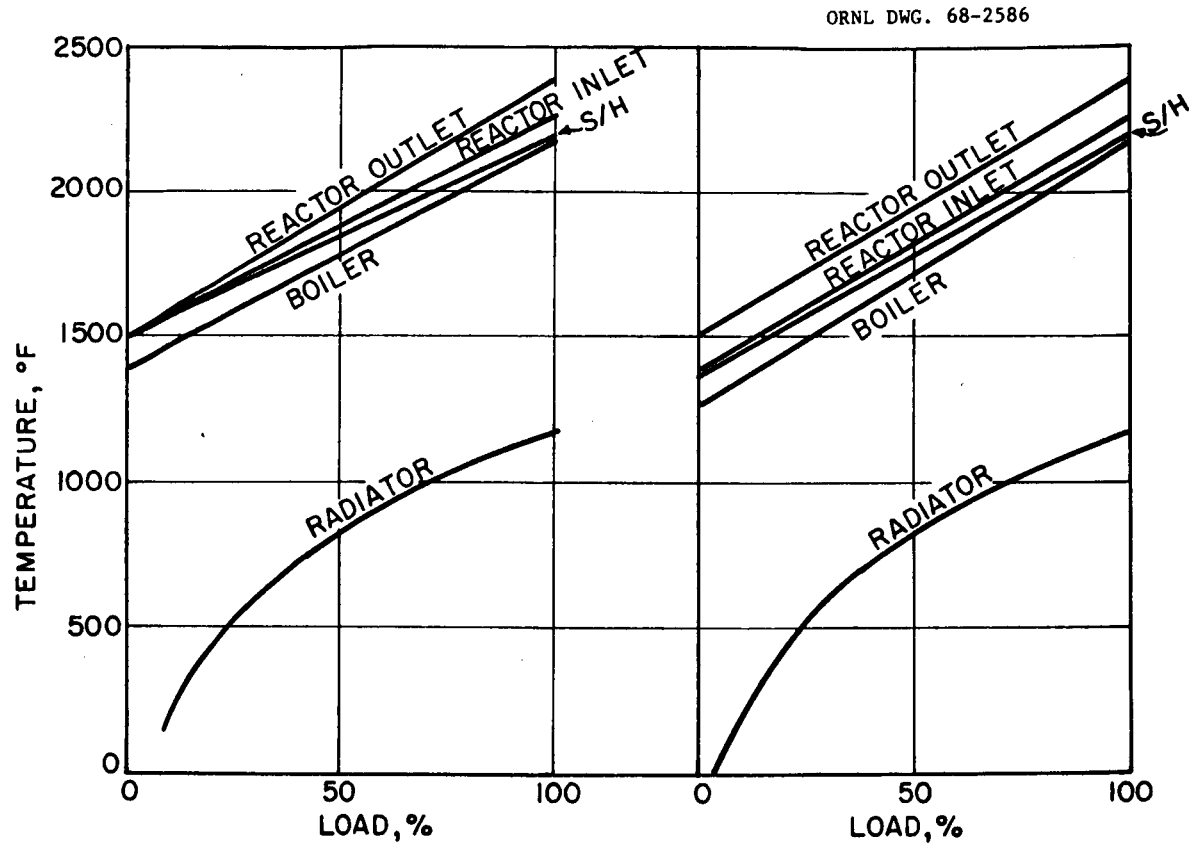


Fig. 14. Effects of Load on System Temperatures in Which the Reactor Circuit Temperature and Flow May Be Controlled.

An additional control to vary the primary circuit flow rate could be added to give the system characteristics shown at the right of Fig. 13. For this system the flow rate would be directly proportional to the load. This would result in only a little improvement in the temperature structure at low loads.

A better approach to the reactor circuit temperature and flow control is shown in Fig. 14. With this arrangement, the reactor outlet temperature would be scheduled to follow a curve roughly parallel to that for the boiler temperature as defined by the turbine operating characteristics indicated in Figs. 10 or 11. There would be considerable latitude in the precise shape of the curve that the reactor outlet temperature would be scheduled to follow, as well as considerable latitude in the precision with which the controls should follow the curve provided that the temperature schedule was chosen to maintain an amount of superheat somewhat greater at part load than at full load as indicated in Fig. 14.

The control scheme having the characteristics shown at the left of Fig. 14 was assumed to have a primary circuit flow rate that would remain constant in order to simplify the control system at the expense of increasing the parasitic load imposed by reactor circuit pump under part-load conditions. The curves at the right of Fig. 14 are for a system in which the primary circuit flow rate would fall off linearly with power to reduce the pumping losses at part load.

While time did not permit detailed studies of the dynamic response of the systems, it is believed that there would be little difference between the system stability and control characteristics that could be obtained with cesium or potassium using the system of either Fig. 13 or Fig. 14.

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